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**Scientific and Technical
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Summary

The performance of 120.65-mm (4.75-in.) bore computer optimized, high-speed design, tapered-roller bearings was investigated at shaft speeds to 20 000 rpm (2.4 million DN or cone-rib tangential velocities to 158 m/sec (31 200 ft/min)). Temperature distribution and bearing heat generation were determined as functions of shaft speed, radial and thrust loads, lubricant flow rate, and lubricant inlet temperature. Lubricant was supplied through holes in the cone at both the large and small ends of the rollers. Test conditions included shaft speeds from 6000 to 20 000 rpm, radial loads from 13 300 to 26 700 N (3000 to 6000 lb), thrust loads from 26 700 to 53 400 N (6000 to 12 000 lb), lubricant flow rates from 0.0038 to 0.0151 m^3/min (1.0 to 4.0 gal/min), and lubricant inlet temperatures of 350 and 364 K (170° and 195° F). The test bearing design was computer optimized for high-speed operation. The lubricant was MIL-L-23699.

The high-speed-design tapered-roller bearing operated successfully at shaft speeds up to 20 000 rpm under heavy thrust and radial loads. Stable temperatures were reached, and no surface distress was observed within the range of conditions tested. Bearing temperatures and heat generation with the high-speed-design bearing were significantly less than those of a modified standard design bearing tested previously.

Bearing temperatures and heat generation increased as expected with increased shaft speed. The highest temperatures measured were at the cup outer surface. With increased lubricant flow rates at both the large and the small ends, bearing temperature decreased, and heat generation increased. A total flow rate as low as 0.0114 m^3/min (3.0 gal/min) gave stable operating temperatures at 20 000 rpm without excessive bearing heat generation. Decreasing oil-in temperature from 364 to 350 K (195° to 170° F) decreased bearing temperature 6 to 14 K (10° to 25° F). Cup cooling was effective in decreasing the high cup temperatures to levels equal to the cone temperature. The effect of load on bearing temperatures was very small relative to the effects of speed and flow rates.

Introduction

Tapered-roller bearings are being used in some helicopter transmissions to carry combined radial, thrust, and moment loads and, in particular, those loads from bevel gears such as high-speed input pinions. For this application tapered-roller bearings have greater load capacity for a given envelope or for a given bearing weight than the more commonly used ball and cylindrical roller bearings. But speed limitations have restricted the use of tapered-roller bearings to lower speed applications compared with ball and cylindrical-roller bearings. The speed limitation is primarily due to the cone-rib-roller-large-end contact, which requires very careful lubrication and cooling consideration for high speed operation. The speed of tapered roller bearings is limited to approximately 0.5 million DN (a cone-rib tangential velocity of approximately 36 m/sec (7000 ft/min)) unless special attention is given to lubricating and designing this cone-rib-roller-end contact. At higher speeds centrifugal effects starve this critical contact of lubricant.

By supplying lubricant directly to this critical contact through holes in the cone, tapered-roller bearings have been successfully operated to 1.81 million DN under heavy combined radial and thrust loads (ref. 1). Under thrust-load-only conditions (ref. 2) speeds as high as 3.5 million DN have been successfully attained.

The internal geometry of the bearings used in these tests (refs. 1 and 2) were standard catalog series designs with modifications of the roller spherical ends and lubrication holes through the cone. Also, some of the bearings used in reference 2 had cages designed for high-speed operation, and the rollers, cones, and cups were made of CBS 1000M material.

The use of computer programs can increase the capability of designing and analyzing tapered-roller bearings for high-speed applications. These programs, described in references 3 and 4, take into account the difficulty of lubricating the contacts in high-speed tapered roller bearings and consider the effects of the elastohydrodynamic (EHD) films in these contacts. The analysis of reference 3 was used to select the internal geometry for the high-speed

design tapered roller bearing used for the research reported herein.

The objective of this research was to determine the operating characteristics, including temperature distribution and heat generation, of a computer optimized, high-speed-design, 120.65-mm (4.75-in.) bore tapered roller bearing at speeds to 20 000 rpm (2.4 million DN). The cone-rib tangential velocity at this speed was 158 m/sec (31 200 ft/min). Independent test variables were shaft speed, radial and thrust loads, lubricant flow rate, and lubricant inlet temperature. The results were compared with the operating characteristics of modified standard test bearings reported in references 1 and 5.

Apparatus and Procedure

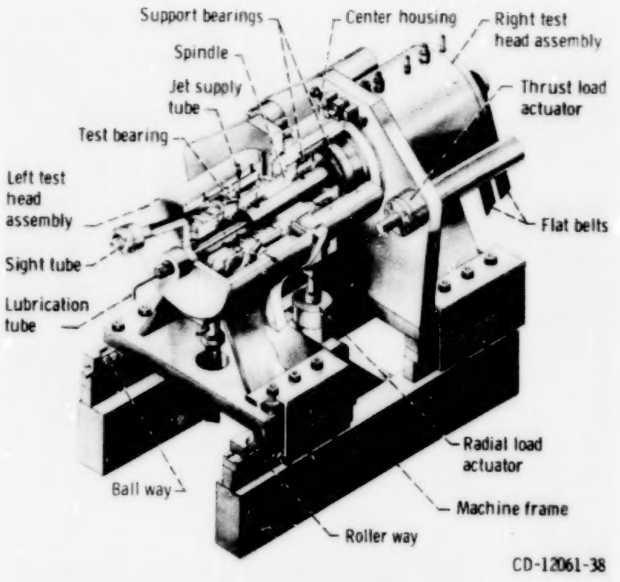
Test Rig

Mechanical arrangement.—The test rig used for this study was that described in reference 1 except for an improved spindle design with closer spacing of the two test bearings. The 75-kilowatt (100-hp) motor was replaced with a 93-kilowatt (125-hp) motor with improved starting torque characteristics required for the higher speed testing. Figure 1 shows both a cutaway drawing and a photograph of the test rig.

The hollow spindle contained annular grooves to distribute the lubricant to radial holes at both the large and small ends of the test bearing (fig. 2). A stationary lubrication tube delivered the desired lubricant flow to the annular grooves. Lubricant flow rates were varied by the varying supply pressures and hole sizes in the lubricant tube. For the tests reported herein jet lubrication was not used. The lubrication system, including pumps, filters, flowmeters, and load actuators, is described in detail in reference 5.

Instrumentation.—Thermocouples were installed for temperature measurements of each test bearing cup, each cylindrical load bearing outer ring, and oil inlet and outlet temperatures on both test and load bearings. Temperatures of the cone bore and cone face of the test bearing on the drive end of the test spindle were measured with thermocouples and transmitted with a frequency modulated (FM) telemetry system.

The test rig vibration level is measured with piezoelectric accelerometers which automatically shut down the test when vibration exceeds a predetermined level because of bearing failures. Proximity probes measure shaft excursion in two planes as well as shaft speed and test bearing separator speed. A power meter was incorporated to monitor test rig power requirements. Preset safety flow switches and oil level switches were used to shut down the test rig in the event of lubrication system malfunction.



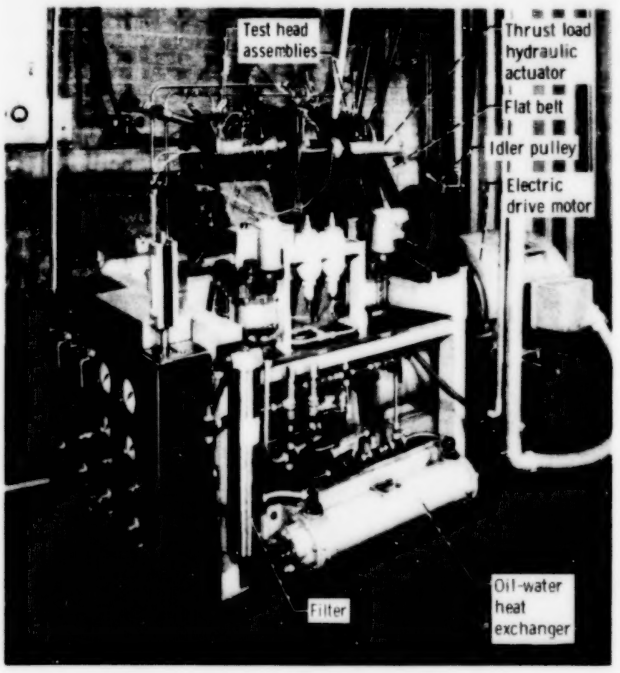


Figure 1. - High-speed tapered roller bearing test rig.

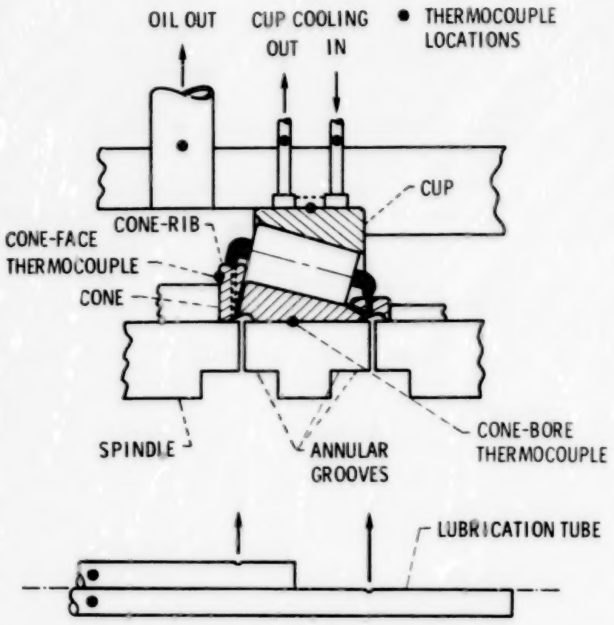


Figure 2. - Test bearing lubrication, cooling, and thermocouple locations.

Test Bearings

The tapered roller test bearings had a bore of 120.65 mm (4.75 in.). The remainder of the bearing internal geometry and external dimensions were selected by computer optimization with due consideration for economical manufacturing using existing tooling.

The computer analysis of reference 3 was used to select optimum cup and roller angles and roller size with a fixed bore of 120.65 mm (4.75 in.) for conditions of a 26 700 N (6000 lb) radial load, a 5 400 N (12 000 lb) thrust load, and a range of shaft speeds from 12 500 to 20 000 rpm. The analysis of reference 3 requires complete definition of a tapered roller bearing and its immediate environment. The optimization process using this analysis consisted of evaluating over 30 selected combinations of cup and roller angles and roller diameters, each defining a unique bearing internal geometry. These combinations were all within the constraints fixed by good design practice for tapered-roller bearings (ref. 3). Criteria for comparison were bearing fatigue life, total bearing heat generation, and cone-rib contact stress and heat generation. Greater consideration was given to those combinations which gave lower stress and heat generation at the cone-rib contact. Further details of the optimization variables are given in the appendix.

The geometry of the computer optimized design is given in table I. This geometry was reviewed by a bearing manufacturer to consider use of existing tooling for more economical manufacturing. As a result, some compromise of the geometry was reached and checked with computer analysis to be satisfactory. This selected geometry is shown in table I, along with the geometry of the standard bearing tested and reported in reference 1.

The rollers in the selected bearing were fully crowned with a crown radius of 25.4 m (1000 in.) and a spherical end radius equal to 80 percent of the apex length. The material of the cup, cone, and rollers was case-carburized, consumable-electrode, vacuum-melted CBS 1000M. The cage was one-piece machined AISI 4340 steel with silver plating and was guided by lands on the cone. The hardnesses, case depth, and surface finishes specifications are shown in table II.

The cone contained 40 1.016-mm (0.040-in.) diameter oil holes at each end. The holes were drilled through from a manifold on the cone bore to the undercuts at each raceway end, as shown in figure 2. In addition, six oil holes were drilled at each end to lubricate the cage land riding surfaces.

The basic dynamic load ratings for this bearing are 70 700 N (15 900 lb) radial load and 51 600 N (11 600 lb) thrust load (the thrust or radial load which gives 10 percent life of 90 million cone revolutions). The Antifriction Bearing Manufacturers Association (AFBMA) basic dynamic capacity is 255 000 N (57 400 lb). This bearing has approximately 10 percent less capacity than the standard bearing of reference 5 because of its optimization of performance at higher speeds.

Lubricant

The oil used for these studies was a 5-centistoke neopentylpolyol (tetra) ester. This type II oil is qualified to MIL-L-23699 as well as to the internal oil specifications of most major aircraft turbine engine manufacturers. Properties of the oil are presented in table III.

Procedure

The test procedure was adjusted according to the test conditions to be evaluated. Generally, a program cycle was defined that would allow the evaluation of a number of conditions without a major interruption. Test parameters such as load, speed, and oil inlet temperature were held constant while the tester was in operation. Lubricant flow rates were adjusted during operation. The test bearings were allowed to reach an equilibrium condition before data were recorded and the next test condition was sought.

TABLE I. - TEST BEARING GEOMETRY

Dimension	Computer optimized bearing	Selected bearing	Standard bearing[a]
Cup half angle	15°	15°53'	17°
Roller half angle	1°30'	1°35'	1°35'
Roller large end diameter, mm (in.)	18.42 (0.725)	18.29 (0.720)	18.29 (0.720)
Number of rollers	22	23	25
Total roller length, mm (in.)	37.31 (1.469)	34.18 (1.3456)	34.17 (1.3452)
Pitch diameter, mm (in.)	155.5 (6.123)	155.1 (6.105)	166.8 (6.569)
Bearing outside diameter, mm (in.)	192.3 (7.571)	190.5 (7.500)	206.4 (8.125)

[a]Bearing used in tests of refs. 1 and 5.

TABLE II. - TEST BEARING SPECIFICATIONS

Cup, cone, and roller material	CBS 1000M
Case hardness, Rockwell C	58 to 64
Core hardness, Rockwell C	25 to 48
Case depth (to 0.5% carbon level after final grind), cm (in.):	
Cup and cone	0.061 to 0.185 (0.024 to 0.073)
Roller	0.091 to 0.201 (0.036 to 0.079)
Surface finish,[a] μm (μin.), rms:	
Cone raceway	0.10 (4)
Cup raceway	0.10 (4)
Cone-rib	0.41 (16)
Roller taper	0.25 (10)
Roller spherical	0.08 (3)

[a]Measured values.

TABLE III. - PROPERTIES OF

TETRAESTER LUBRICANT

Additives . . . Antiwear, oxidation inhibitor, antifoam	
Kinematic viscosity, cS, at -	
311 K (100° F)	28.5
372 K (210° F)	5.22
477 K (400° F)	1.31
Flash point, K (°F)	533 (500)
Autoignition temperature, K (°F)	694 (800)
Pour point, K (°F)	214 (-75)
Volatility (6.5 hr at 477 K (400° F)), wt.%	3.2
Specific heat at 372 K (210° F),	
J/kg K (Btu/lb °F)	2140 (0.493)
Thermal conductivity at 372 K (210° F),	
J/m sec K (Btu/hr ft °F)	0.15 (0.088)
Specific gravity at 372 K (210° F)	0.931

Results and Discussion

Effect of Lubricant Flow on Bearing Temperatures

The effect of lubricant flow rate was determined for a variety of speeds, loads, and oil-in temperatures with the computer optimized, high-speed tapered-roller bearing. Lubricant was supplied through holes at both the small and large ends of the cone as shown in figure 2. No jet lubrication was used. Temperatures of the 120.65-mm (4.75-in.) bore tapered roller test bearing at the drive end of the test spindle were measured on the cone bore and the cone face as well as on the outer surface of the cup. Oil-out temperatures were also measured. Test spindle speeds ranged from 6000 to 20 000 rpm. Thrust loads varied from 26 700 to 53 400 N (6000 to 12 000 lb). Radial loads varied from 13 300 to 26 700 N (3000 to 6000 lb). The calculated maximum Hertz stresses at several of these conditions are shown in table IV. Total lubricant flow rates varied from 0.0038 to 0.0151 m^3/min (1.0 to 4.0 gal/min).

Test bearing temperatures and oil-out temperatures, measured at these conditions, are shown in figures 3 to 6. Figures 3 and 4 show very little effect of radial or thrust load on cone-face temperature. These data are typical throughout the range of variables; that is, regardless of speed, oil-in temperature, or flow rates, load had little effect on bearing or oil-out temperature. Therefore, nearly all

TABLE IV. - CALCULATED MAXIMUM HERTZ STRESSES AT SEVERAL CONDITIONS OF HIGH-SPEED TAPERED ROLLER BEARING TESTS

[Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).]

Shaft speed, rpm	Maximum Hertz stress at most heavily loaded roller					
	Cone		Cup		Cone-rib	
	MPa	psi	MPa	psi	MPa	psi
10 000	1290	187 000	1230	178 000	195	28 300
15 000	1190	173 000	1230	178 000	218	31 600
20 000	1080	157 000	1230	178 000	239	34 700

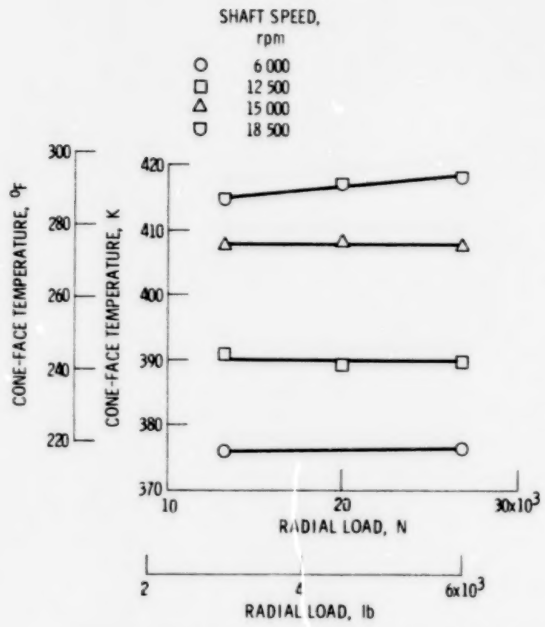


Figure 3. - Effect of radial load on cone-face temperature. Thrust load, 53 400 N (12 000 lb); oil-in temperature, 364 K (195° F); total oil flow, 0.0076 m^3/min (2.0 gal/min).

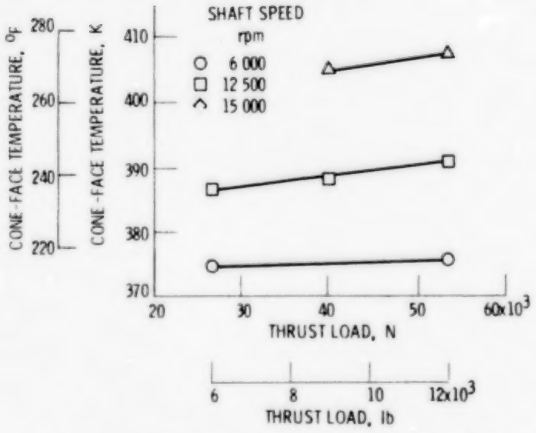
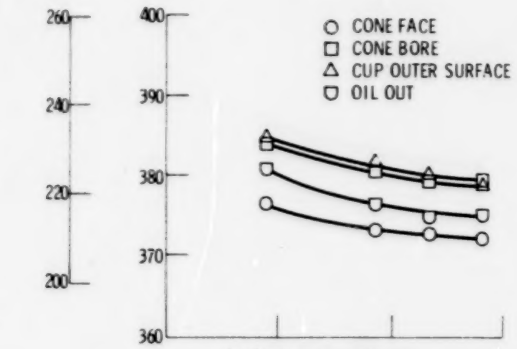
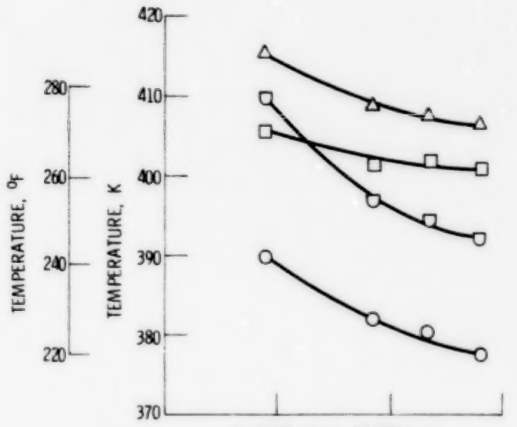


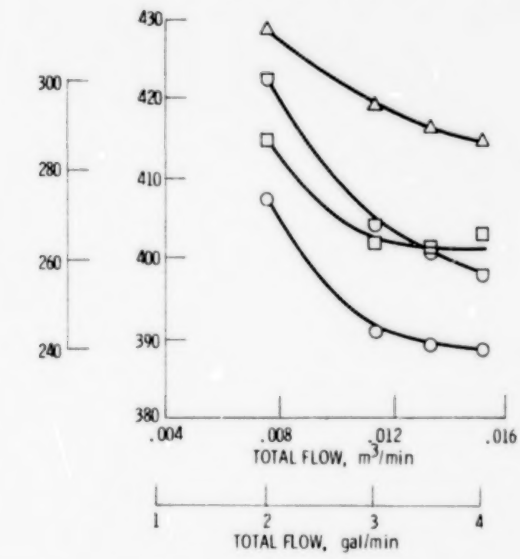
Figure 4. - Effect of thrust load on cone-face temperature. Radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); total oil flow, 0.0076 m^3/min (2.0 gal/min).



(a) Shaft speed, 6000 rpm.

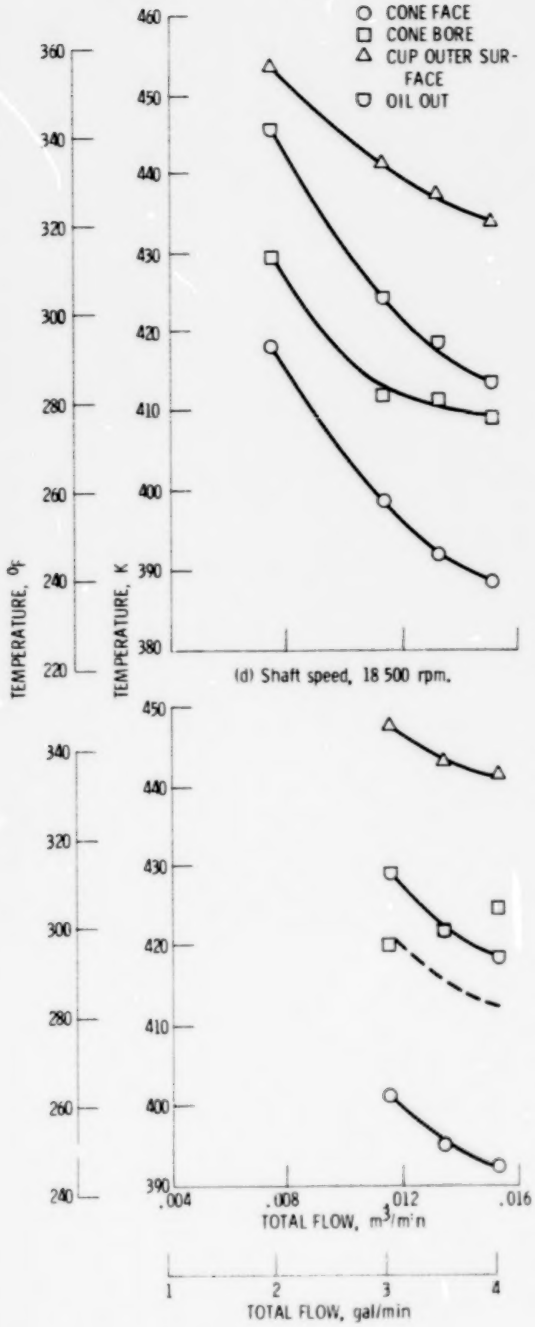


(b) Shaft speed, 12 500 rpm.



(c) Shaft speed, 15 000 rpm.

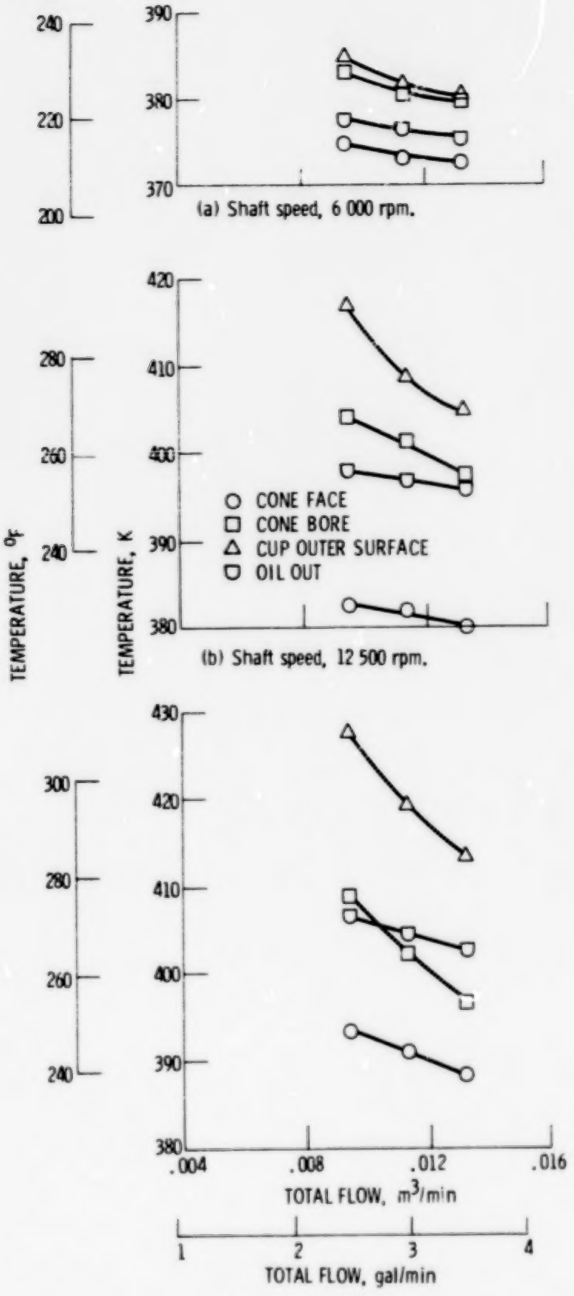
Figure 5. - Temperature as a function of total flow rate with variable large end flow. Small end flow rate, 0.0038 m^3/min (1.0 gal/min); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F).



(d) Shaft speed, 18 500 rpm.

(e) Shaft speed, 20 000 rpm.

Figure 5. - Concluded.

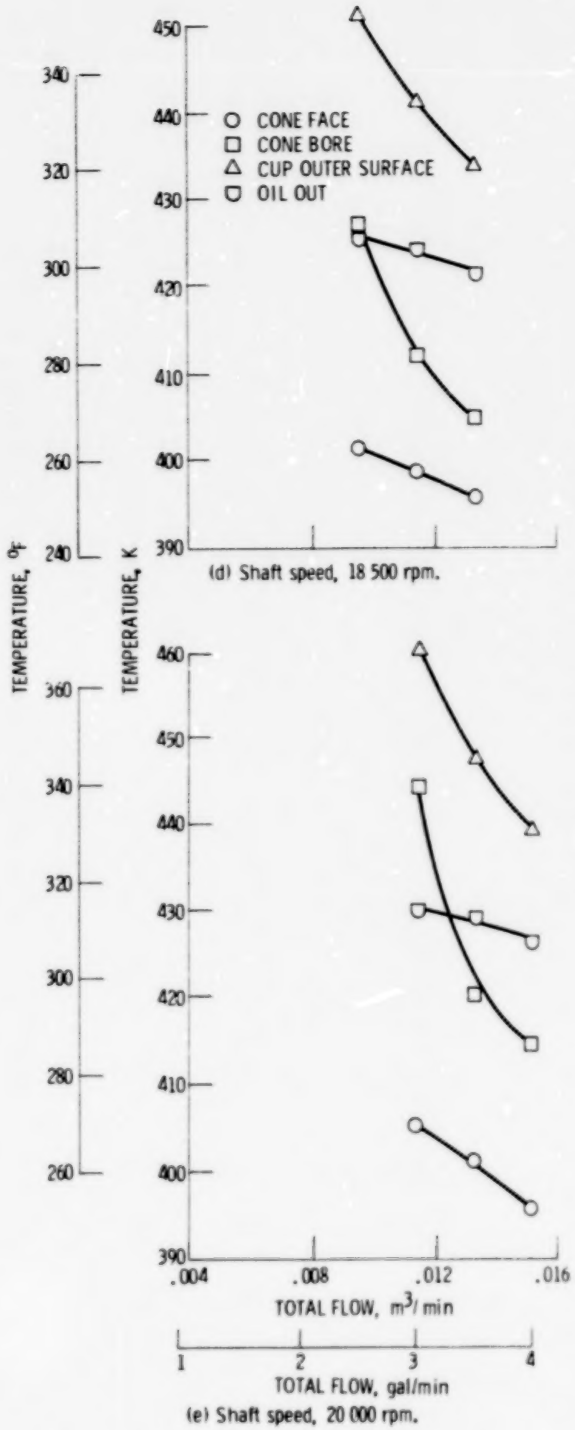


(a) Shaft speed, 6 000 rpm.

(b) Shaft speed, 12 500 rpm.

(c) Shaft speed, 15 000 rpm.

Figure 6. - Temperature as a function of total flow rate with variable small end flow. Large end flow rate, 0.0076 m^3/min (2.0 gal/min); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6 000 lb); oil-in temperature, 364 K (195° F).



(d) Shaft speed, 18 500 rpm.

(e) Shaft speed, 20 000 rpm.

Figure 6. - Concluded.

the data shown in the remainder of this report are for one load condition, namely, 53 400-N (12 000-lb) thrust load and 26 700-N (6000-lb) radial load.

Figure 5 shows the general decrease in bearing and oil-out temperatures with increased lubricant flow rate to the large end with a constant 0.0038 m^3/min (1.0 gal/min) to the small end. At the higher speeds, for example, 18 500 rpm in figure 5(d), the effect of the lubricant flow rate is much greater than at the lower speeds. Bearing temperatures at 18 500 rpm are decreased as much as 28 K (50° F) as total flow rate is increased from 0.0076 to 0.0151 m^3/min (2.0 to 4.0 gal/min). The temperature of the cup outer surface is greater than the cone bore or cone face throughout the range of flow rate and speed conditions.

Figure 6 shows the effect of increasing flow rate to the small end of the bearing from 0.0019 to 0.0057 m^3/min (0.5 to 1.5 gal/min) with a constant 0.0076 m^3/min (2.0 gal/min) at the large end. The effects of this were greater at the higher speeds. A comparison of figures 5 and 6 shows that small-end flow rate has a greater effect on cup temperature than on cone-face temperature and that the opposite is true for large-end flow rate. This effect should be expected since the lubricant introduced at the small end is pumped through the bearing to the large end and in doing so is thrown centrifugally to the cup raceway surface, thus cooling the cup. Lubricant introduced through the large end cools the cone rib and quickly exits the bearing with less cooling effect on the cup. The large effect of small-end flow rate on cone-bore temperature at the higher speeds (figs. 6(d) and (e)) indicates its effectiveness in cooling the cone.

It is apparent that, for speeds in the range of 15 000 to 20 000 rpm, flow rates of 0.0076 m^3/min (2.0 gal/min) at the large end and 0.0038 m^3/min (1.0 gal/min) at the small end are near optimum for this design and size of bearing. Smaller flow rates give significantly higher temperatures, and large flow rates only increase bearing power loss as will be shown later.

Effect of Shaft Speed

The effect of shaft speed on bearing and oil-out temperatures is shown in figure 7 for a total flow rate of 0.0114 m^3/min (3.0 gal/min). The greater effect of speed on cup outer surface temperature than on cone-bore temperature is readily apparent. As will be shown later, cup cooling can be used to decrease the high cup temperatures at the higher speeds. It should be noted that the oil-out temperature increases with speed at about the same rate as the cup outer-surface temperature. Such a great increase in temperature with speed should not be attributed to greater centrifugal loading of the rollers on the cup since load has been shown to have a minor effect on

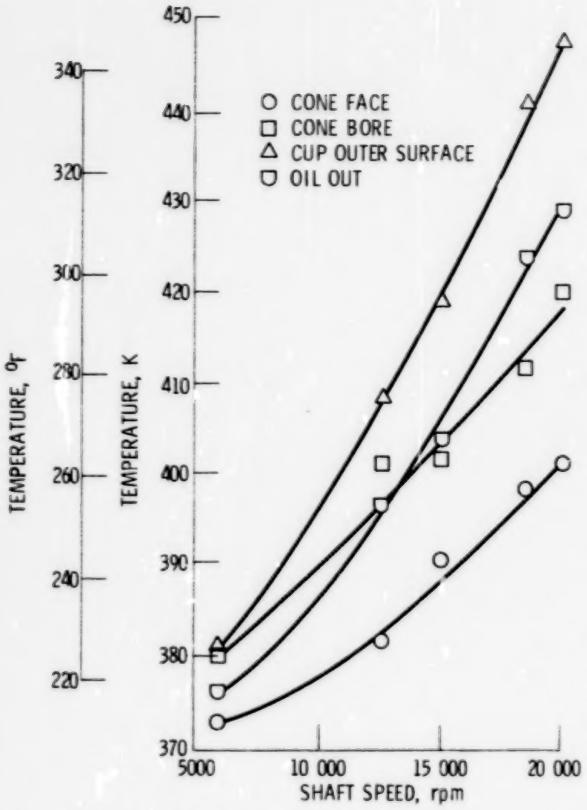
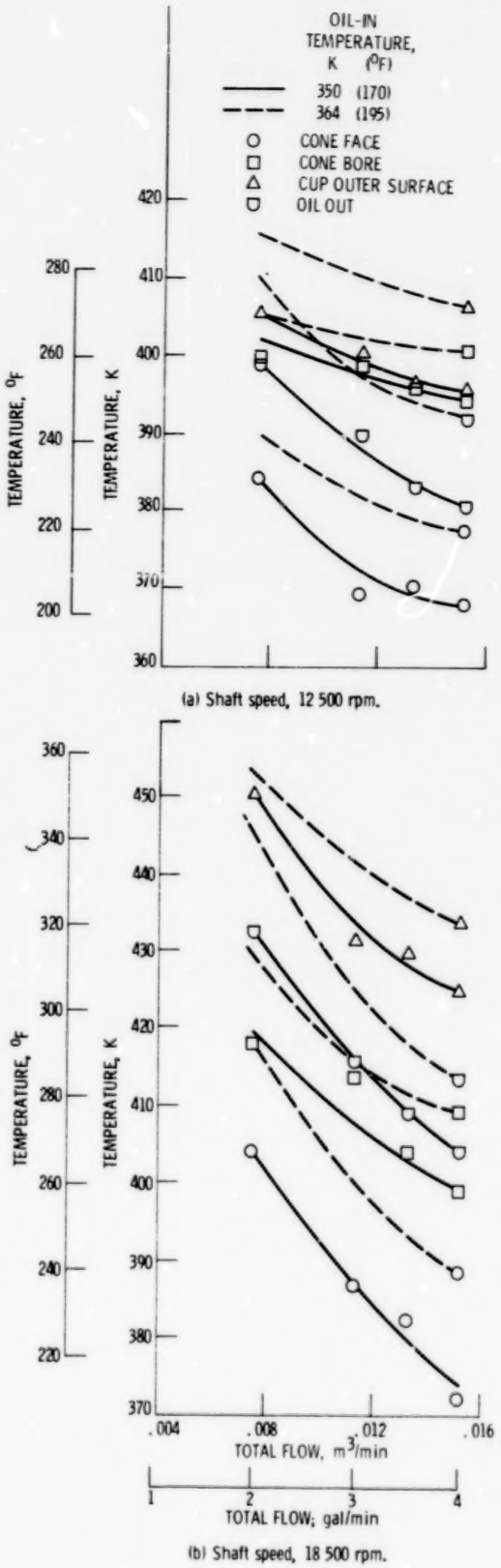


Figure 7. - Effect of shaft speed on bearing and oil-out temperatures. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); large end flow rate, 0.0076 m^3/min (2.0 gal/min) small end flow rate, 0.0038 m^3/min (1.0 gal/min).

temperature. Also, since the cage is guided on lands on the cone, greater heat generation at those guiding surfaces at higher speeds is expected, along with higher cone temperature. Apparently, the lubricant flow through the holes in the cone (both large and small ends) very adequately cools the cone, and by the time it reaches the cup raceway, it has been heated appreciably and has less cooling capacity.

The test bearing cage speed was measured at all shaft speed conditions. The ratio of cage speed to shaft speed did not vary more than 1 percent from a nominal value of 0.445 throughout the range of speed, load, and lubricant flow conditions.

Stable operating temperatures were achieved at all speed, load, and lubricant flow rate conditions. Some of the lower lubricant flow rate points were not attempted at 20 000 rpm due to anticipated excessive temperatures. Observations of the bearings after tests at 20 000 rpm showed no signs of surface distress on the roller-raceway or roller/cone-rib contacting surfaces. Likewise, cage contacting surfaces showed no abnormal wear or distress.



(a) Shaft speed, 12 500 rpm.

(b) Shaft speed, 18 500 rpm.

Figure 8. - Effect of oil-in temperature on bearing and oil-out temperatures. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); small end flow rate, 0.0038 m^3/min (1.0 gal/min).

Effect of Oil-In Temperature

The effect of oil-in temperature on bearing and oil-out temperatures is shown in figure 8. In figure 8(a) temperatures at an oil-in temperature of 350 K (170° F) and 12 500 rpm shaft speed are shown compared with data from figure 5(b) at a 364 K (195° F) oil-in temperature. At the lower oil-in temperature, bearing and oil-out temperatures generally were 6 to 14 K (10° to 18° F) less than at the higher oil-in temperature. Least affected was the cone bore temperature.

In figure 8(b) similar data are shown for a shaft speed of 18 500 rpm. Here the temperatures were generally 6 to 14 K (10° to 25° F) lower for the lower oil-in temperatures and least affected was the cup outer surface temperature. Also, the cone-bore temperature was decreased the full 14 K (25° F) of the decrease in oil-in temperature. Generally, for both shaft speeds, the bearing and oil-out temperatures were affected slightly more by oil-in temperature at the higher flow rates.

Effect of Cup Cooling

As was shown in figure 7, the highest bearing temperatures measured were at the cup outer surface, and the difference from cone temperatures was increasingly greater at higher speeds. The higher cup temperatures may be decreased with the use of cup cooling oil flowing in contact with the outer surface of the cup (as shown in fig. 2). Figure 9 shows the effects of cup cooling flows up to 0.0028 m^3/min (0.75 gal/min) at 12 500 and 18 500 rpm and a 350 K (170° F) oil-in temperature. As expected, greatest effects were at the cup outer surface where temperatures were decreased 11 K (20° F) with a cup cooling flow of 0.0019 m^3/min (0.5 gal/min) at 18 500 rpm. Higher cooling flows had little or no effect.

Limited data were obtained at 18 500 rpm and an oil-in temperature of 364 K (195° F). As shown in table V, a cup cooling flow rate of 0.0038 m^3/min (1.0 gal/min) was sufficient to lower the cup outer surface 14 K (25 deg F) or to a level equal to the cone-bore temperature.

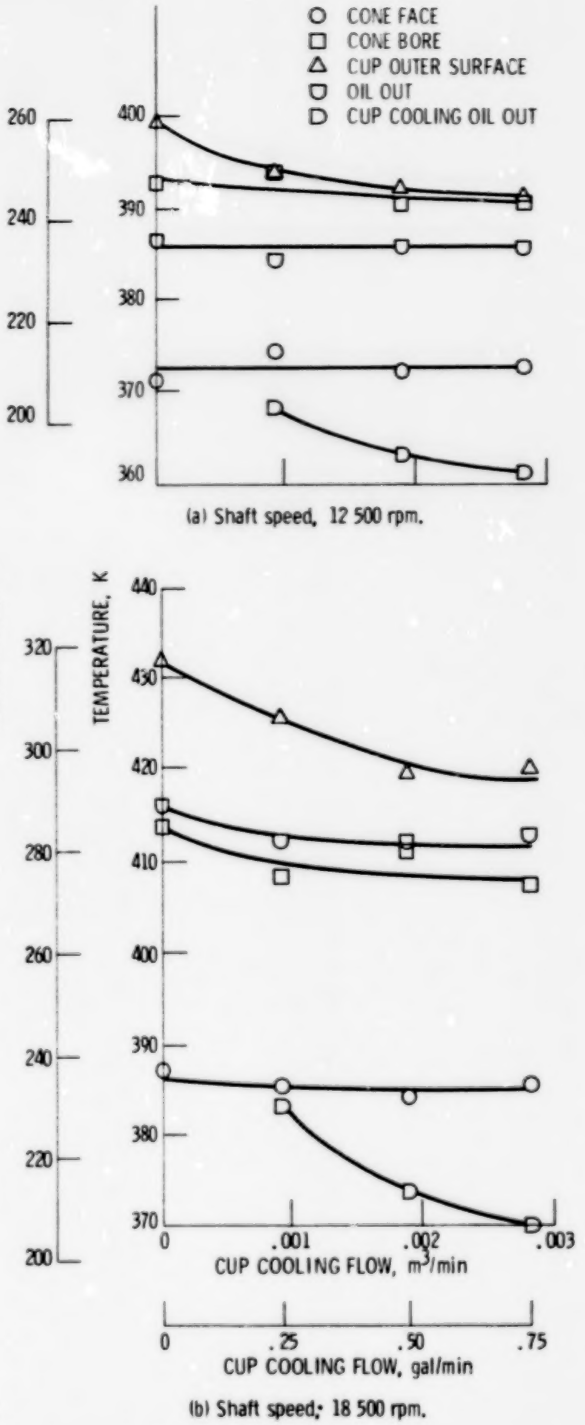


Figure 9. - Effect of cup cooling flow rate on bearing and oil-out temperatures. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); large end flow rate, 0.0076 m^3/min (2.0 gal/min); small end flow rate, 0.0038 m^3/min (1.0 gal/min); oil-in temperature, 350 K (195° F).

Bearing Power Loss

The power loss from the bearing is dissipated in the form of heat by conduction to the lubricant and by convection and radiation to the surrounding environment. Lubricant outlet temperature from the bearing was measured for all conditions of flow. Heat

TABLE V. - EFFECT OF CUP COOLING ON BEARING AND OIL-OUT TEMPERATURES

[Shaft speed, 18 500 rpm; thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); large-end flow, 0.0076 m^3/min (2.0 gal/min); small-end flow, 0.0038 m^3/min (1.0 gal/min).]

Cup cooling flow rate, m^3/min (gpm)	Cone face	Cone bore	Cup outer surface	Oil-out	Cup cooling oil-out
	Temperature, K (°F)				
0	389 (240)	423 (302)	438 (329)	426 (307)	--------
0.0038 (1.0)	391 (245)	424 (303)	424 (304)	426 (308)	386 (235)

transferred to the lubricant was calculated using the following standard heat transfer equation:

$$Q_T = MC_p(t_{out} - t_{in}) \qquad (1)$$

where

Q_T total heat transfer to lubricant, J/min (Btu/min)
M lubricant mass flow, kg/min (lb/min)
C_p specific heat, J/kg K (Btu/lb °F)
t_{out} oil outlet temperature, K (°F)
t_{in} oil inlet temperature, K (°F)

The result of these heat-transfer calculations is shown in figure 10 as a function of total flow rate. (For convenience, heat values are converted from J/min to kW.) The heat transferred to the lubricant increases with increased lubricant flow rate. The effect is greater at higher speeds.

Figure 11 shows an even greater effect of shaft speed on heat transferred to the lubricant. The effect of shaft speed can be closely approximated by the relation $Q_T \propto N^{1.35}$ where N is the shaft speed in rpm. These increases are expected because of the increased lubricant drag or churning. These heat quantities are a portion of the heat generated in the test bearings and do not include heat transferred from the bearing by conduction, convection, and radiation. At higher bearing temperatures the heat transferred by these latter forms should become a greater portion of the total.

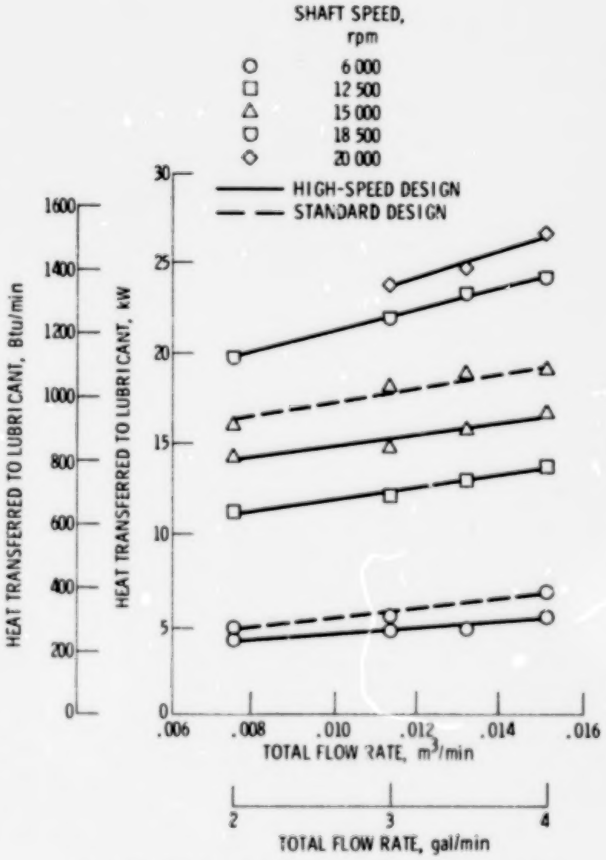


Figure 10. - Heat transferred to lubricant as function of total flow rate. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); small end flow rate, 0.0038 m^3/min (1.0 gal/min).

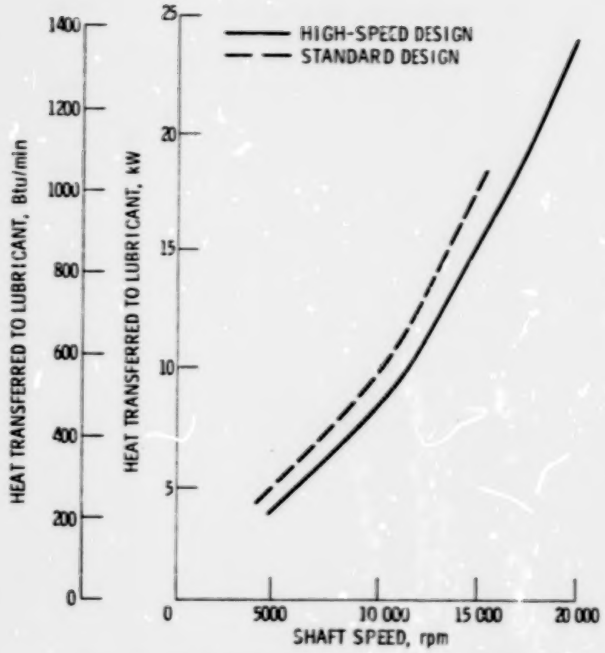


Figure 11. - Heat transferred to lubricant as function of shaft speed. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); large end flow rate, 0.0076 m^3/min (2.0 gal/min); small end flow rate, 0.0038 m^3/min (1.0 gal/min).

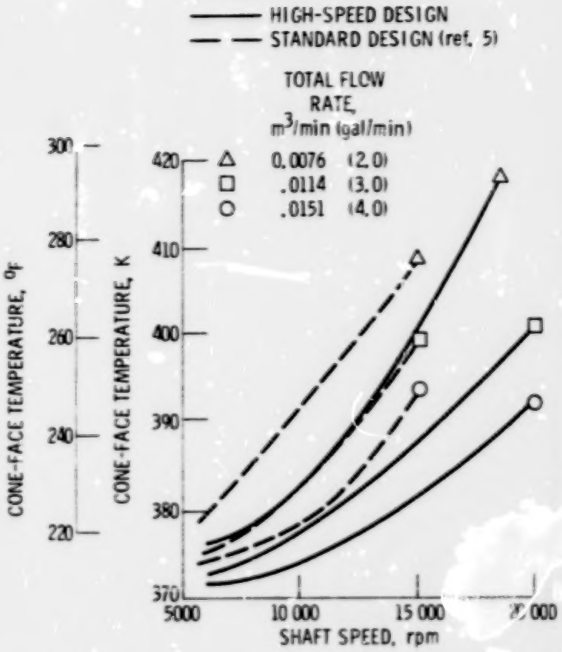


Figure 12. - Cone face temperatures for high-speed design compared with those for standard design. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); small end flow rate, 0.0038 m^3/min (1.0 gal/min).

Comparison with Standard Design Bearing

In references 1 and 5 data from tests with a modified standard catalog series tapered-roller bearing of the same bore (120.65 mm (4.75 in.)) were presented for speeds up to 15 000 rpm. Loads and total flow rates were over the same range as those reported herein for the high-speed design bearing. Major differences in the two bearing designs were smaller cup angle, smaller pitch and outside diameters, and fewer rollers in the high-speed design (table I). The cage of the high-speed design bearing was made to be guided by lands on the cone; whereas, the standard bearing had a stamped steel cage made to be guided by the rollers and was of lesser strength for high-speed operation.

Lubricant to the large end of the standard bearing was directed through holes in the cone to the cone-rib surface; lubricant to the small end was through a pair of jets directed at the small end of the rollers. Comparisons herein were made with equal flow rates for both bearings at both the large end and the small end. At the small end, a constant 0.0038 m^3/min (1.0 gal/min) was used, and that was fed through jets for the standard design and through the cone small end for the high-speed design.

Materials with higher temperature capabilities were used for the high-speed design bearings: Cups, cones, and rollers were made of CBS-1000M, and the cage was AISI 4340. The standard design bearing, on the other hand, had AISI 4320 cups, cones, and rollers and AISI 1010 cages.

In figure 12, cone-face temperatures of the computer optimized, high-speed design from figure 5 are compared with data from reference 5. The symbols are used only to identify the curves at each flow rate. At 15 000 rpm, the high-speed-design bearing operates 8 to 11 K (15° to 20° F) cooler than the standard design. The improvement at 6000 rpm, however, is much less. The cup outer surface temperature is also lower for the high-speed-design bearing as shown in figure 13, although the improvement is slightly less than that at the cone face.

Also shown in figure 10 are data for the standard design bearing showing heat transferred to the lubricant at 6000 and 15 000 rpm. The high-speed-design bearing has lower heat generation, as represented by heat transferred to the lubricant, than the standard design bearing at both speeds. As shown in figure 11, the improvement with the high-speed design is approximately 16 percent throughout the range of 6000 to 15 000 rpm.

Summary of Results

The performance of 120.65-mm (4.75-in.) bore, computer-optimized, high-speed-design, tapered-

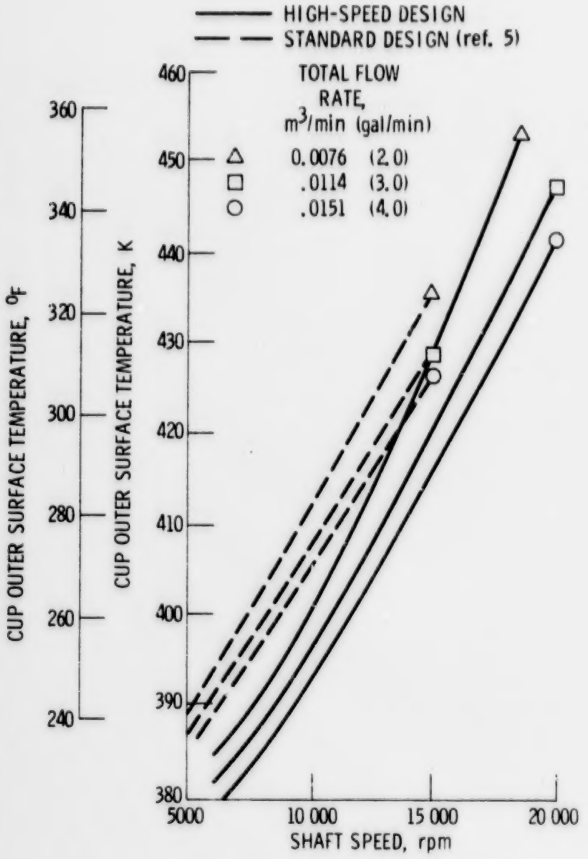


Figure 13. - Cup outer surface temperatures for high-speed design compared with those for standard design. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); small end flow rate, 0.0038 m^3/min (1.0 gal/min).

roller bearings was investigated at shaft speeds up to 20 000 rpm (cone-rib tangential velocities up to 158 m/sec (31 200 ft/min)). Temperature distribution and bearing heat generation were determined as a function of shaft speed, radial and thrust loads, lubricant flow rate, and lubricant inlet temperature. Lubricant was supplied by holes through the cone at both the large and the small ends of the roller. Test conditions included shaft speeds from 6000 to 20 000 rpm, radial loads from 13 300 to 26 700 N (3000 to 6000 lb), thrust loads from 26 700 to 53 400 N (6000 to 12 000 lb), lubricant flow rates from 0.0038 to 0.0151 m^3/min (1.0 to 4.0 gal/min), and lubricant inlet temperatures of 350 and 364 K (170° and 195° F). The following results were obtained:

1. The computer-optimized, high-speed-design, tapered-roller bearing operated successfully at shaft speeds up to 20 000 rpm under heavy thrust and radial loads. Stable temperatures were reached, and no surface distress was observed.

2. Bearing temperatures and heat generation with the high-speed-design bearing were significantly less than those of a modified standard bearing tested previously.

3. Bearing temperatures and heat generation increased with increased shaft speed. The highest temperatures measured were at the cup outer surface.

4. With increased lubricant flow rates (either large end or small end), bearing temperatures decreased and heat generation increased. A total flow rate as low as 0.0114 m^3/min (3.0 gal/min) gave stable operating temperatures at 20 000 rpm without excessive bearing heat generation.

5. Decreasing oil-in temperature from 364 K (195° F) to 350 K (170° F) decreased bearing temperatures 6 to 14 K (10° to 25° F).

6. Cup cooling was effective in decreasing the high cup temperature to levels equal to the cone temperature.

7. The effect of load on bearing temperatures was very small compared with speed and lubricant flow rate effects.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, March 4, 1980,
505-04.

Appendix—Optimization Variables

The computer optimization utilized the program described in reference 3. The bearing bore diameter was fixed at 120.65 mm (4.75 in.). Over 30 cases were evaluated using cup half angles α_o of 12°, 15° and 18°; roller half angles ν of 1.0°, 1.5° and 2.0°; and roller large end diameters ranging from 13.3 to 27.9 mm (0.525 to 1.100 in.).

Other dimensions which were fixed by good design practice (ref. 3) were

$$A = 5.08 \text{ mm } (0.20 \text{ in.})$$

$$B = 5.08 \text{ mm } (0.20 \text{ in.})$$

$$C = 0.15 \text{ W}$$

$$E = 0.635 \text{ mm } (0.025 \text{ in.})$$

$$R_S = 0.8\ L_A$$

where R_S is roller spherical end radius and L_A is apex length,

$$T_w = 0.2\ D_m$$

where T_W is cage web thickness, and D_m is roller mean diameter,

$$h = 2.54 \text{ mm } (0.10 \text{ in.})$$

where h is roller end-flange contact height. And

$$1.0 < L/D < 2.6$$

where L is total roller length and D is roller large end diameter. These dimensions are defined in figure 14.

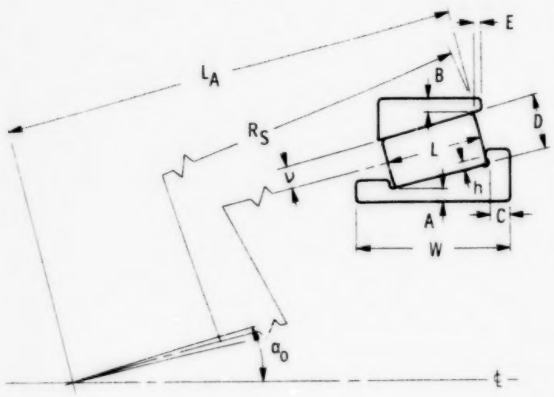


Figure 14. - Tapered-roller bearing geometry.

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15. Supplementary Notes

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Stanley I. Pinel and Hans R. Signer: Industrial Tectonics, Inc., Compton, California.

16. Abstract

The performance of 120.65-mm (4.75-in.) bore high-speed-design, tapered-roller bearings was investigated at shaft speeds to 20 000 rpm (2.4 million DN) under combined thrust and radial load. The test bearing design was computer optimized for high-speed operation. Temperature distribution and bearing heat generation were determined as a function of shaft speed, radial and thrust loads, lubricant flow rates, and lubricant inlet temperature. The high-speed-design, tapered-roller bearing operated successfully at shaft speeds up to 20 000 rpm under heavy thrust and radial loads. Bearing temperatures and heat generation with the high-speed-design bearing were significantly less than those of a modified standard bearing tested previously. Cup cooling was effective in decreasing the high cup temperatures to levels equal to the cone temperature.

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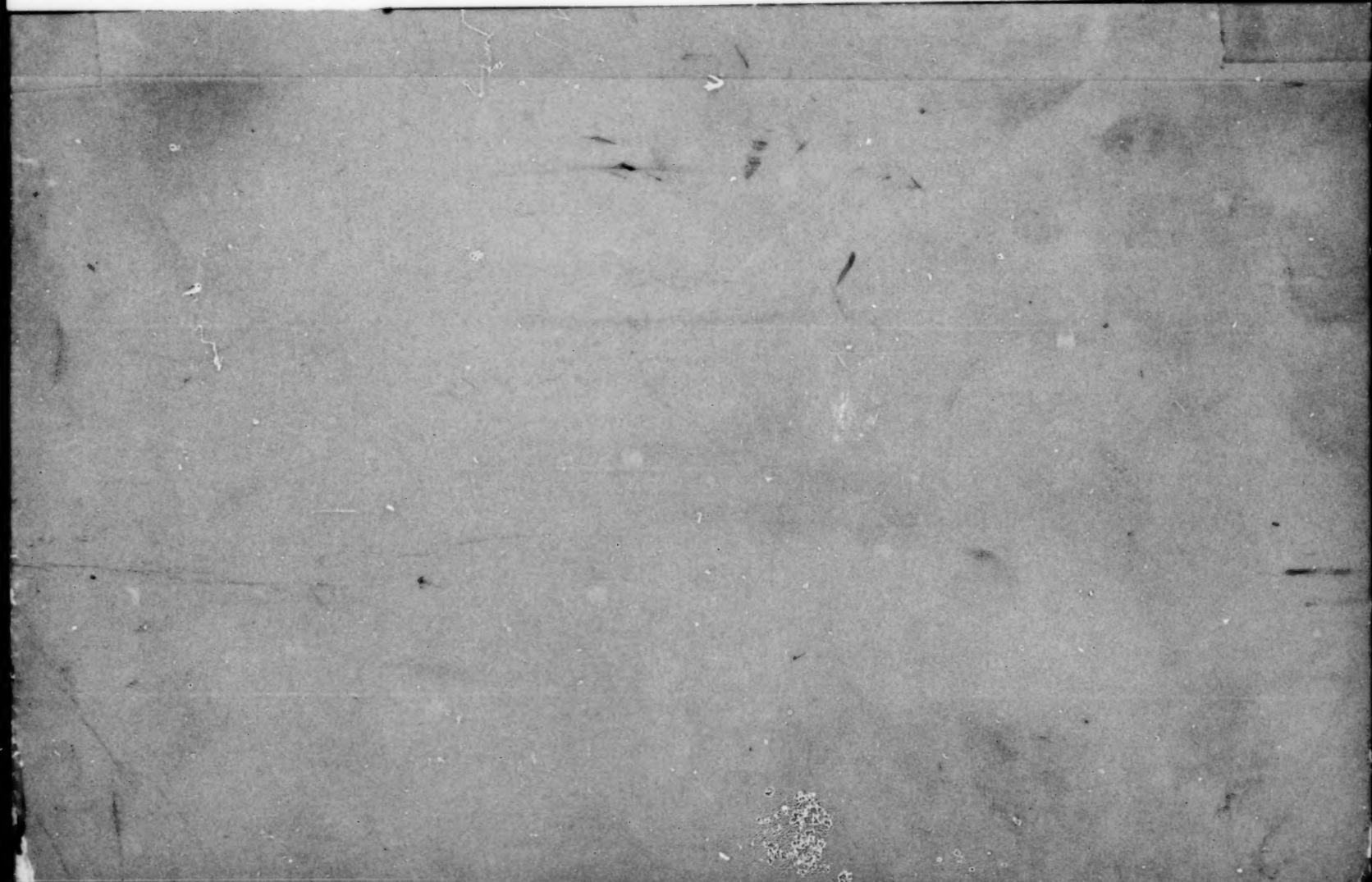

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